

# Comparison of Two Total Energy Systems for a Diesel Power Generation Plant

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*This article compares the capabilities and limitations, as well as the associated costs for two total energy systems for a diesel power generation plant. Both systems utilize waste heat from engine cooling water and waste heat from exhaust gases. Pressurized water heat recovery system is simple in nature, requires no engine modifications, but operates at lower temperature ranges. On the other hand, a two-phase ebullient system operates the engine at constant temperature, provides higher temperature water or steam to the load, but is more expensive.*

## I. Introduction

Total energy systems have been widely studied as a more effective means of using the depletable energy sources. Waste heat from the diesel engine represents a potential source of energy that can be further utilized for heating and/or cooling purposes or, coupled with a Rankine bottoming cycle, to produce extra power. Diesel engines for power generation have an efficiency between 30 to 35 percent, which means that 65 to 70 percent is wasted to the atmosphere.

First, about 40 percent of the total waste heat is exchanged during the combustion process in the engine, which tends to increase the cylinder wall temperature to such an extent that without cooling, the lubrication oil can evaporate resulting in engine damage. Water is usually employed to cool the engine. The temperature of the cooling water leaving the engines range from 87.7°C (190°F) to 104.4°C (220°F). However, to avoid excessive thermal stresses in the engine, it is advisable to have a temperature difference of 5.55°C (10°F) to 8.33°C (15°F) between the inlet and the outlet water temperatures. The hot cooling water from the engine is then fed into an air-cooled or a water-cooled radiator

before recirculating it back to the engine. Instead of rejecting the heat to the atmosphere, as much as 90 to 100 percent can be recovered for heating or cooling purposes. Second, the other 60 percent of the waste heat is in the form of sensible heat in the exhaust gases at the end of the expansion stroke. The temperature of the exhaust gases varies from engine to engine, ranging between 371.1°C (700°F) and 698.9°C (1200°F). The sensible heat in the exhaust gases can be extracted by means of a stack-type heat exchanger. To avoid water condensate in the exhaust ducts, the temperature of the exhaust gases leaving the stack cannot be lower than 176.7° (350°F), which means that only 60 to 70 percent of the waste heat from exhaust gases is recoverable.

DSN overseas stations in Spain and Australia operate their diesel engines continuously around the clock to provide power for the stations' needs. Therefore, it is felt necessary to make use of the waste heat from the engines to conserve the valuable and depletable diesel fuel. DSS 62 is currently using the diesel waste heat for heating and cooling. DSS 61/63 is in the process of installing a waste heat recovery system for space heating, and an effort is underway to further use the diesel waste heat to provide cooling.

## II. Relationship of Engine Load and the Available Waste Heat

To utilize the waste heat from diesel engines for heating and/or cooling, it is important to investigate the engine operating loads and the energy required for heating and/or cooling, since they determine the sufficiency or deficiency of the engine waste heat. By employing the waste heat for heating and cooling, the engine will be operating at a reduced load level since it no longer provides the electricity needed for the present heating and cooling devices such as electric boilers and vapor compression refrigeration units. In other words, the waste heat available will be less. On the other hand, the coefficient of performance (COP) of the absorption chiller and the efficiency of the heat exchangers should be taken into consideration to determine the heat needed to meet the heating and/or cooling load. For instance, a diesel engine operating at 500 kWe is providing heating and cooling loads of 150 kWt and 100 kWt, respectively. Utilizing the waste heat for heating and cooling, the engine load should be reduced by 200 kWe assuming an efficiency of 0.9 for the electric boiler and the COP of 3.0 for the vapor compression chiller. Further, taking the COP of the absorption chiller at 0.6 and the heat exchanger efficiency at 0.9, the heat needed will be enhanced to approximately 350 kWt, and obviously, the waste heat from jacket water alone for an engine operating at 300 kWe load is not sufficient and the waste heat from exhaust gases will have to be used to compensate for the heat needed. Figure 1 shows the relationship between the engine load and the sufficiency of jacket waste heat for a one-day period. From hour 12 to hour 22, the waste heat from jacket water alone can no longer provide enough heat for both heating and cooling; the heat available from exhaust gases or back-up systems should be further utilized to make up for the deficiency.

In the event that the waste heat from exhaust gases is used to supplement the balance of heat required, several heat recovery systems can be used and their capabilities and limitations, as well as their associated costs, should be carefully investigated.

## III. Heat Recovery Systems

Basically, there are two types of heat recovery systems that utilize both the engine jacket water and the exhaust gases: 1) a pressurized water system, and 2) an ebullient or 2-phase steam/water system. The pressurized hot-water system operates the engines at a temperature range of 82.2°C (180°F) to 92.9°C (210°F). One-third of the fuel energy input to the engine will dissipate into the jacket water, which would cause a temperature difference between the water coming into the engine and the water leaving the engines of

not more than 8.33°C (15°F). To achieve this, a temperature-controlled water flow rate should be maintained. The water leaving the engine is then fed to a gas-heat recovery unit to extract the waste heat from the exhaust gases. The gas-heat recovery units are typical gas-to-water heat exchangers with exhaust gases running in the hot side, and the engine jacket water in the cold side. Reference 1 lists the common temperatures of exhaust gases for various models of caterpillar engines. Since the temperature of the exhaust gases leaving the heat recovery units should not be lower than 176.7°C (350°F), the amount of recoverable waste heat can easily be estimated. Moreover, the temperature of the water leaving the heat recovery units also can be determined. The temperature increase for the water across the heat recovery units will generally not exceed 5.55°C (10°F). As a result, a maximum water temperature of approximately 100°C (212°F) can be achieved only by using this scheme. For engines operating at temperatures close to 100°C (212°F), pressurized water systems should be employed throughout the cooling water loop to avoid flashing in the engines. It is also advisable to use in the external loop a heat exchanger to deliver the heat to the load to form a closed loop for the engine cooling water. A booster pump will also be installed since the existing engine pump will not be enough to circulate the water. At times when cooling is not needed, the heat can be removed using the existing radiators or converted to chilled water and stored in a chilled water storage tank. The pressurized hot-water system does not require any modifications to the existing engines. However, the maximum obtainable hot-water temperature (100°C) is lower than the absorption chiller manufacturer's specification (116°C), which means an oversized chiller would be used at a lower capacity rather than the nominal size, even though it operates with the same coefficient of performance (COP). Table 1 lists the absorption chillers with the nominal sizes at 115.6°C (240°F) and the degraded capacities at 90.5°C (195°F). Figure 2 depicts the schematic flow diagram for the pressurized hot-water system.

The 2-phase steam/water ebullient system removes the heat from the engines by heat of vaporization. For an ebullient system, the engines are operating at a constant high temperature ranging between 112.8°C (235°F) to 121.1°C (250°F). The cooling water is fed to the engines at the operating temperature. As the water passes through the cylinders, the heat rejected will vaporize a portion of the water into wet steam. The wet steam is fed to heat recovery units which serve both as mufflers and as steam separators. Using the heat from the exhaust gases will further convert the wet steam to dry saturated or superheated steam, which feeds the absorption chiller. Theoretically, the cooling water into the engine should have the same temperature as the water leaving the engine, but a 1.1°C (2°F) to 1.6°C (3°F) increase in temperature is normally experienced. Because of the smaller temperature

gradient across the engine, the engine is subject to smaller thermal stresses, which give the engines a longer life. Further, the cooling water circulation is by natural convection and a pump is not necessary. The water-steam flow rate is about 1/70 of that of the pressurized system due to the large latent heat of water. However, this system requires a separate cooling loop for oil coolers, which uses water at 29.4°C (85°F). The cooling water loop should maintain a pressure of 69 kPa (10 psi) to 103.5 kPa (15 psi) to avoid flashing inside the engines; the temperature of the steam collected will be in the neighborhood of 115.6°C (240°F) to 121.1°C (250°F), which satisfies the chiller manufacturer's specification. For engines running at such a high temperature, the seals, plumbing system, and even the cylinder head will have to be replaced for high-temperature applications. Thus, it is appropriate to convert the engines to the ebullient cooling system during a major overhaul period. When excess steam is not needed, the steam will be condensed in a bypass condensing unit and returned to the heat recovery units. Figure 3 shows the schematic diagram of a high-temperature ebullient system. Water treatment for the engine cooling water is necessary for an ebullient system, and it is critical to engine operation. It is desirable to have chemical testing of the water every other day to keep the water at a certain pH level.

#### IV. Cost Comparison

A pressurized hot-water system does not require any engine modification. However, the system needs a heat exchanger, a booster pump, and an oversized absorption chiller. On the other hand, an ebullient system requires no heat exchanger, works with nominal size absorption chiller, but requires engine modification, an excess steam condensing unit, a holding tank for the condensate, and a more expensive heat recovery unit. It is necessary to have a thorough breakdown of the costs for each system to select the most cost-effective system.

To demonstrate the cost comparison of the two systems, assume a diesel power generation plant that operates four diesel engines and utilizes the waste heat from the engines to feed an absorbing chiller of 700-kWt capacity.

Table 2 lists the cost for each component for both systems. The installation cost and the maintenance cost for the two systems are assumed the same. The main differences in cost between the two systems fall into three categories: 1) engine modification, 2) heat recovery units, and 3) the absorption chillers. For ebullient systems, engines have to be modified for high-temperature application. The cost of the modification is taken as \$7,000 per engine including the replacement of the cylinder head, seals and necessary changes in the plumbing system. The pressurized hot-water system uses a gas-water heat exchanger for the heat recovery unit, while the ebullient system employs a more expensive heat recovery unit acting as both a heat exchanger and a steam separator. Further, the pressurized hot-water system requires an oversized chiller to provide the same amount of cooling as the ebullient system. The material cost for hot-water system and the ebullient system are approximately \$103K and \$150K, respectively.

#### V. Conclusion and Remarks

A pressurized hot-water system and the high-temperature ebullient system are the two commonly used heat recovery systems associated with diesel total energy systems. Both systems recover waste heat from the engine jacket water as well as waste heat from the exhaust gases. The pressurized hot water system is simple, requires no engine modification, but needs an oversized absorption chiller. On the contrary, high-temperature ebullient systems require engine modifications, more expensive heat recovery units and a water treatment system. The installation and maintenance costs for both systems are very much the same, but the material cost for the ebullient systems are 1.5 times higher than that for the pressurized hot-water system. Furthermore, for future increase in station cooling consumption, the pressurized hot-water system has the capability of providing the additional cooling with a supplementary source such as solar energy, to enhance the hot water temperature to, in turn, increase the capacity of the absorption chiller, independent of the engine waste heat.

#### Reference

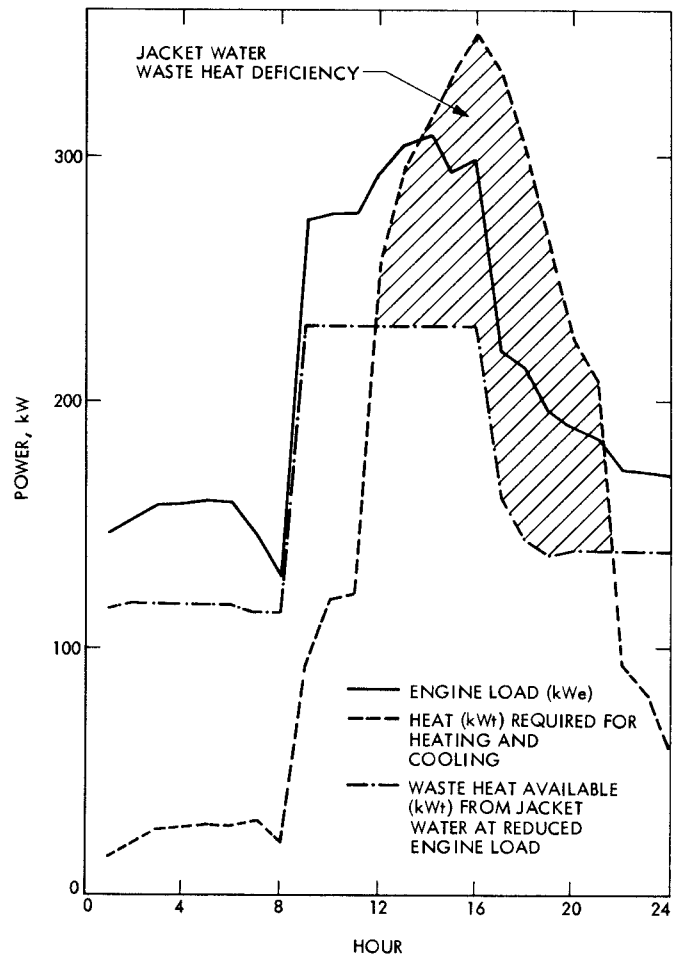
1. *Air Conditioning Handbook*. Caterpillar Tractor Company, Peoria, Illinois, 1970.

**Table 1. Degradation of chiller capacities and corresponding costs**

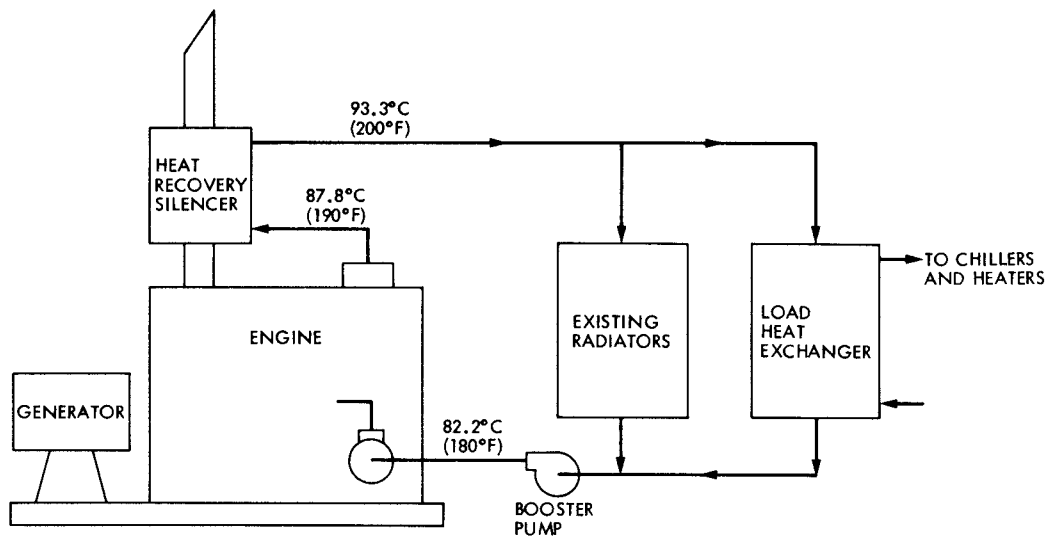
Nominal size at 115.6°C (240°F), kWt	Operation capacity at 90.5°C (195°F), kWt	Estimated cost, \$K
2180	879	70.0
1976	791	64.5
1835	738	61.0
1723	703	59.0
1396	562	48.5
1097	440	42.0
819	334	36.0

**Table 2. Cost comparison for pressurized hot-water system and high-temperature ebullient system**

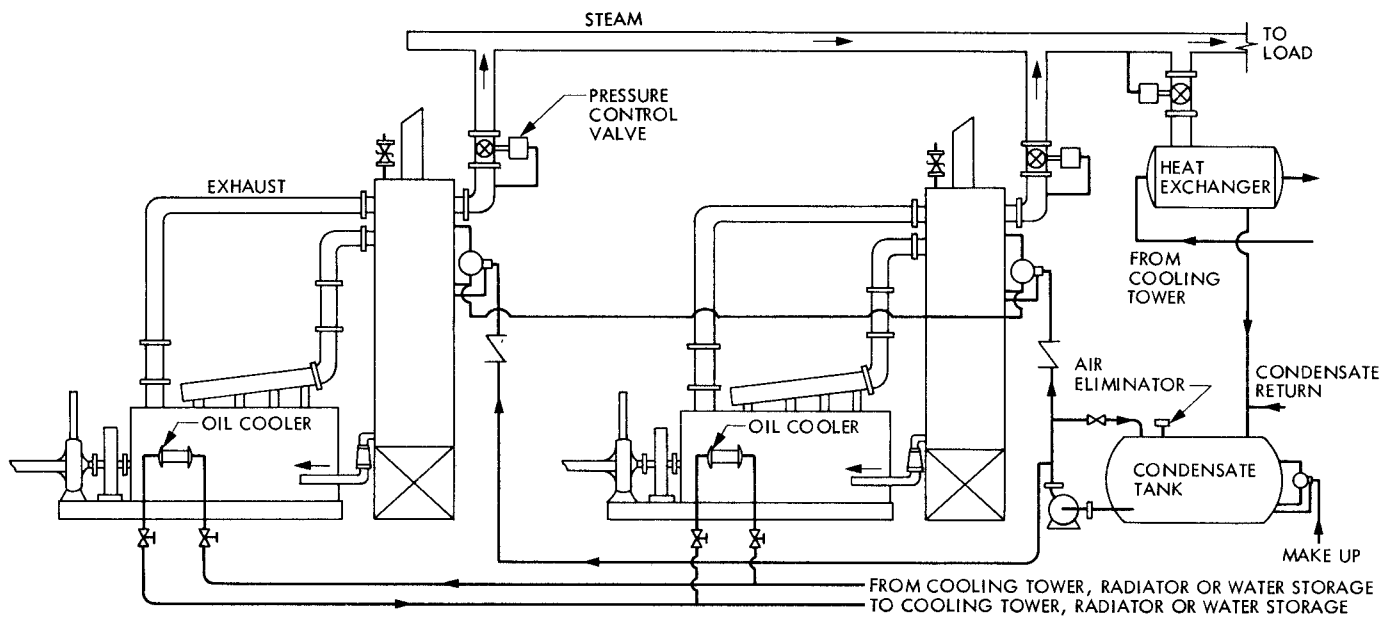
Component	Pressurized hot-water system, \$K	High-temperature ebullient system, \$K
Engine modification	—	@ 7 × 4 = 28
Heat recovery units and Insulation and controls	@10 × 4 = 40	@20 × 4 = 80
Heat exchanger or condensing unit	4	10
Absorption chiller and accessories	59	36
Total Cost	103	154



**Fig. 1. Example of waste heat profile for one day**



**Fig. 2. Pressurized hot-water system**



**Fig. 3. High-temperature 2-phase water/steam ebullient system**